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Investigation of the effects of steam injection on performance and NO emissions of a diesel engine running with ethanol–diesel blend

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ABSTRACT

The use of ethanol–diesel blends in diesel engines without any modifications negatively affects the engine performance and NOx emissions. However, steam injection method decreases NOx emissions and improves the engine performance. In this study, steam injection method is applied into a single cylinder, four-stroke, direct injection, naturally aspirated diesel engine fueled with ethanol–diesel blend in order improve the performance and NOx emissions by using two-zone combustion model for 15% ethanol addition and 20% steam ratios at full load condition. The results obtained are compared with conventional diesel engine (D), steam injected diesel engine (D + S20), diesel engine fueled with ethanol–diesel blend (E15) and steam injected diesel engine fueled with ethanol–diesel blend (E15 + S20) in terms of performance and NO emissions. The results showed that as NO emissions considerably decrease the performance significantly increases with steam injection method.

1. Introduction

Recently, researchers focused on using of alternative fuels in diesel engines due to depletion of reserves based on petroleum fuels and assertive environmental restrictions with regard to decreasing of pollutant emissions. Ethanol is one of the alternative alcoholic fuel for diesel engines [1]. Additionally, it is renewable fuel having higher miscibility and it could be obtained from plants such as barley, sugar cane, sugar beets, corn and sorghum. Hence, the attention of the use of ethanol in diesel engines has rapidly increased, in recent years [2–4].

In order to use the ethanol in diesel engines, various techniques were developed such as blending [4–10], emulsification [11–13], fumigation [14–18] and dual injection [19,20]. Separate injection systems are required for each fuel in dual injection systems. In the fumigation systems, secondary fuel is added to the intake air. Therefore, setting up these fuel delivery systems in diesel engines can be pricy and hard technically. Ethanol is mixed with diesel before injection in the blending method; if an emulsifier is used to blend the fuels in order to avoid separation then it is called emulsion method.

The blending and emulsion methods are more advantageous thanks to absence of technical modifications and the ease of application. However, little amounts of ethanol could be added to diesel engines in order to get fine consequences due to the limited miscibility, reductions in cetane number, viscosity and heating value [10,21,22].

So many studies have been done on the performance and emission characteristics of diesel engines running with ethanol–diesel mixtures. Can et al. [1] performed a study on influences of ethanol–diesel mixtures in rate of 10% and 15% of fuel volume on performance and emissions of a indirect injection diesel engine with different injection pressures (150, 200 and 250 bar). It was observed that as CO, SO_2, soot emissions, torque and power decrease, NOx emissions considerably increase with ethanol addition.

Sayin [23] conducted an experimental investigation on the performance and exhaust emissions of a direct injection diesel engine with running methanol–diesel and ethanol–diesel mixtures at 5% and 10% of diesel fuel by volume. Test engine was run between 1000 and 1800 rpm at 30 N m torque constantly. Experimental consequences demonstrated that brake thermal efficiency, CO, total hydrocarbon, smoke emissions reduced as NOx emissions raised with the fuel mixtures. In another study, Sayin and Canakci [24] investigated the effects of injection timing and mixture rates on the engine performance and exhaust emissions of a single cylinder diesel engine. The experiments were performed using five different injection timings from 21 CA to 33 CA BTDC with increasing of 3 CA and three mixture rates were chosen from 0% to 15% with increasing of 5%. It was shown that NOx and CO_2 emissions raised while break thermal efficiency, CO and HC reduced with increasing ethanol rates in the mixture. For the retarded injection timings, NOx and CO_2 emissions decreased and HC and CO emissions enhanced. For the advanced injection timings, NOx and CO_2 emissions increased and HC and CO emissions decreased. Break

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Nomenclature

A	heat transfer area (cm^2)
C_v	constant volume specific heat ($J\,g^{-1}\,K^{-1}$)
C_p	constant pressure specific heat ($J\,g^{-1}\,K^{-1}$)
C	blow by coefficient
B	bore (cm)
F	fuel–air ratio
h	specific enthalpy ($J\,g^{-1}$)
h_{tr}	heat transfer coefficient ($W\,m^{-2}\,K^{-1}$)
H	enthalpy ($J\,g^{-1}$)
H_u	low heat value ($J\,g^{-1}$)
m	mass (g)
$\dot{m}$	time-dependent mass rate ($g\,s^{-1}$)
M	molecular weight
n	injection constant
P	pressure (bar)
R	gas constant ($J\,g^{-1}\,K^{-1}$)
NR	revolution per minute
NY	total mole number
Q	loss heat passed through the cylinder wall (J)
$\dot{Q}$	rate of heat transfer (W)
RGF	residual gas fraction
s	specific entropy ($J\,g^{-1}\,K^{-1}$)
S	stroke (cm)
$\bar{S}_p$	mean piston velocity ($m\,s^{-1}$)
T	temperature (K)
u	specific internal energy ($J\,g^{-1}$)
v	specific volume ($cm^3\,g^{-1}$)
V	volume (cm^3)
W	work output (J)
x	burn fraction
$\dot{x}_i$	fraction rate of the total injected fuel mass
$K_{\%}$	ratio of the steam mass to the fuel mass

Greek letters

α	atomic number of carbon for diesel fuel
β	atomic number of hydrogen for diesel fuel
γ	atomic number of oxygen for diesel fuel
δ	atomic number of nitrogen for diesel fuel
ε	ratio of half stroke to rod length, molar fuel–air ratio
ϕ	equivalence ratio
$\Gamma(n)$	gamma function
θ	crank angle (°)
ρ	mass density ($kg\,m^{-3}$)
τ	time (ms)
ω	angular velocity ($rad\,s^{-1}$)

Subscripts

0	at the beginning of compression for steam injected condition
1	at the beginning of compression for standard condition
a	air, atomic number of carbon for secondary fuel
b	burned zone, atomic number of hydrogen for secondary fuel
c	atomic number of oxygen for secondary fuel
d	diesel fuel, atomic number of nitrogen for secondary fuel
cyl	cylinder
di	injection duration parameter
dif	diffusive combustion phase
f	fuel
fi	injected fuel
id	ignition delay (ms)
l	leak, loss
pre	premixed combustion phase
r	reference
sf	secondary fuel
si	start of fuel injection (°)
st	stoichiometric
ste	steam
u	unburned zone
w	cylinder walls

thermal efficiency decreased at all retarded and advanced injection timings as compared to the original injection timing.

Rakopoulos et al. [25] performed an experimental investigation on the effects of ethanol–diesel blends on the performance and emissions of a turbocharged, DI diesel engine using 5% and 10% ethanol rates by volume in diesel fuel. The consequences of the experimental tests showed that the use of ethanol mixtures reduced CO and NOx emissions while increased THC emissions and break thermal efficiency.

Huang et al. [26] carried out a study on the use of the ethanol–diesel blends in a diesel engine. In the study, it could be seen that brake thermal efficiencies reduced depending on raising ethanol in the fuel mixtures. The smoke emissions from the engine running with ethanol-blends were lower than that running with pure diesel at all conditions. CO decreased at half and higher loads and increased lower loads and engine speeds. HC emissions decreased at peak loads and high speeds. NOx emissions were varied with respect to engine speeds loads and blends.

Abu-Qudais et al. [4] performed a study to define the optimum percentages of ethanol in the fumigation and blending methods in terms of performance and emissions. The results showed that the optimum percentage for ethanol fumigation and blending are 20% and 15%, respectively.

Rakopoulos et al. [27] performed an experimental study to investigate the effects of using of ethanol in diesel fuel mixtures with 5%, 10% and 15% (by volume) on the performance and emissions of a standard high-speed, direct injection diesel engine. The results of the study showed that soot, NOx, CO reduced as break thermal efficiency and HC emission increased.

Ajav et al. [28] studied on the influences of ethanol–diesel blends on the performance and emissions of a stationary diesel engine using of 5%, 10%, 15% and 20% ethanol–diesel blends. The results demonstrated that brake power, break thermal efficiency, CO and NOx emissions of the engine fueled ethanol–diesel blends decreased compared to that fueled pure diesel.

Based on previous literature reviews, the engine performance and NOx emissions could be got worsen with the addition of the ethanol into diesel fuel used in the diesel engines [1,23,24,26–30]. In order to reduce these adverse effects of the ethanol mixtures on diesel engines some techniques can be used such as EGR, water and steam injection into combustion chamber [31–38]. Even though, EGR is commonly used to minimize NOx emissions, performance is dramatically decreased [31–33]. Another known NOx reduction technique is water injection into the engine cylinder with different methods [34,35]. On the other hand, one of the substantial handicaps of water injection techniques is

that condensed water in the cylinder downgrades the quality of lubrication oil and raises the attrition rate of moving parts of the engine [38]. Recently, one of the developed techniques so as to reduce NOx emissions and improve the engine performance is steam injection into suction manifold [38–41]. Gonca et al. [36,37] conducted a theoretical study on steam injected diesel engine and miller cycled diesel engine using two-zone combustion model. The results of the study showed that steam injection could minimize NOx emissions from diesel engines and improve effective efficiency and effective power. Parlak et al. [38] developed an electronically controlled steam injection method for a diesel engine. When this technique was implemented to diesel engine, NOx emissions decreased up to 33%, effective power and torque increase up to 3% and SFC minimize up to 5% at full load tests. Cesur et al. [39] and Kokkulunk et al. [40] applied electronically controlled steam injection system into gasoline and diesel engines and they obtained similar results to those of Parlak et al.'s study [38]. In these studies, the optimum steam ratio was determined as 20% of injected fuel by mass in terms of NOx reduction and performance improvement. Therefore, steam injection method could be implemented to a diesel engine fueled ethanol–diesel mixture so as to improve performance and reduce NOx emissions. Gonca et al. [41] defined optimum steam temperatures and optimum steam mass ratios for turbocharged internal combustion engines.

In this study, apart from the prior studies, the effects of steam injection into a diesel engine fueled with ethanol–diesel blend (15% by volume) have been studied in terms of performance and NO emissions using a two-zone combustion model, which could be used for any combination of dual fuel blends, developed by Gonca et al. [36,37,42]. In the literature there is not any study on the diesel engine operating with ethanol–diesel mixture and steam injection system. Hence, this study has a considerable novelty and originality.

2. Theoretical model

Combustion simulation of diesel engine fueled ethanol–diesel blend with and without steam injection is carried out by using two-zone combustion model in order to calculate NO emissions, effective efficiency and power [36,37,42]. The burnt and unburnt gas zones are divided by region border.

The fuel injected into the engine cylinder reacts with the air in the unburnt region and then occurs a part of the burnt gas zone by combustion. In the combustion chamber, the equation of the energy conservation in differential form is written as following:

$$m\frac{du}{d\theta} + u\frac{dm}{d\theta} = -\frac{dQ_b}{d\theta} - \frac{dQ_u}{d\theta} - P\frac{dV}{d\theta} + \frac{dm_{fi}}{d\theta}h_{fi} - \frac{dm_l}{d\theta}h_l \tag{1}$$

where m_l is leak mass and m_{fi} is mass of injected fuel; h_{fi} and h_l are enthalpies of injected fuel and leak mass per cycle, respectively. The first term of the left side of the equation is the internal energy rate and the second term is the mass rate depending on crank angle. The heat transfer rates from burnt and unburnt zone are given, respectively as:

$$\dot{Q}_b = h_{tr}A_bT_{bw} \tag{2}$$

$$\dot{Q}_u = h_{tr}A_uT_{uw} \tag{3}$$

where $T_{uw} = T_u - T_w$ and $T_{bw} = T_b - T_w$, h_{tr} is heat transfer coefficient of gas regions, A_b and A_u are the areas of burnt and unburnt gas inside the cylinder which are in contact with the cylinder walls and T_b, T_u and T_w are the temperatures of the burnt, unburnt gas zones and cylinder walls [43]. The change of stroke volume depending on crank angle is given as following:

$$\frac{dV}{d\theta} = \frac{\pi}{8}B^2S\ \mathrm{Sin}\theta\left[1 + \varepsilon\frac{\cos\theta}{\left(1-\varepsilon^2\sin^2\theta\right)^{\frac{1}{2}}}\right] \tag{4}$$

So as to solve the differential equations, the following expressions are used in the theoretical model. Internal energy:

$$\frac{du}{d\theta} = C_p - \frac{pv}{T}\left(\frac{\partial\ln v}{\partial\ln T}\right)_p\frac{dT}{d\theta} - v\left[\frac{\partial\ln v}{\partial\ln T} + \frac{\partial\ln v}{\partial\ln P}\right]_T\frac{dp}{d\theta} + \frac{\partial u}{\partial\phi}\frac{d\phi}{d\theta} \tag{5}$$

The burnt gas leaking through the rings:

$$\frac{dm_l}{d\theta} = \frac{Cm}{\omega} \tag{6}$$

where C and ω are blow by coefficient and angular velocity, respectively. The mass balance inside the cylinder can be expressed as:

$$m = m_a + m_{fi} \tag{7}$$

where m_a and m_{fi} are the masses of the air and injected fuel respectively. If Eq. (7) is written in differential form, it becomes:

$$\frac{dm}{d\theta} = \frac{dm_a}{d\theta} + \frac{dm_{fi}}{d\theta} \tag{8}$$

The air and injected fuel flow rates depending on crank angle within the cylinder could be stated respectively as:

$$\frac{dm_a}{d\theta} = \frac{-\dot{m}_l/\omega}{1+\phi F_{st}} = \frac{-Cm_a}{\omega} \tag{9}$$

$$\frac{dm_{fi}}{d\theta} = \frac{1}{\omega}\left(\dot{m}_{fi} - \frac{\dot{m}_l\phi F_{st}}{1+\phi F_{st}}\right) = \frac{\dot{m}_{fi} - Cm_{fi}}{\omega} \tag{10}$$

where $\dot{m}_l$, ϕ and F_{st} are the time-dependent gas leak rate, equivalence ratio and stoichiometric fuel–air ratio by mass, respectively. $\dot{m}_{fi}$ is the rate of time-dependent injected fuel and could be given as:

$$\dot{m}_{fi} = \dot{x}_i m_{fi} \tag{11}$$

where m_{fi} and $\dot{x}_i$ are the total mass of the fuel to be injected and fraction rate of the total injected fuel mass respectively, which could be expressed as:

$$m_{fi} = \phi F_{st}(1 - \mathrm{RGF})m_a \tag{12}$$

$$\dot{x}_i = \frac{\omega}{\theta_{di}\Gamma(n)}\left(\frac{\theta-\theta_{si}}{\theta_{di}}\right)^{n-1}\exp\left[\frac{-(\theta-\theta_{si})}{\theta_{di}}\right] \tag{13}$$

where $\Gamma(n)$ is the gamma function [43], θ_{di} is a parameter of injection duration and θ_{si} is the start angle of fuel injection. The gamma function is given as:

$$\ln\Gamma(n) = \left(n-\frac{1}{2}\right)\ln(n) - n + \frac{1}{2}\ln(2\pi) + \frac{1}{12n} - \frac{1}{360n^3} + \frac{1}{1260n^5} - \frac{1}{1680n^7} \tag{14}$$

The values of n could be taken for the diesel engine with open chamber as $1 \leqslant n \leqslant 2$ and for close chamber as $3 \leqslant n \leqslant 5$ but precise value is dependent on fuel used and engine design [43]. Differential equations solved in the calculation of the processes that consist during the period from the beginning of the compression to the end of the expansion process are given in Eqs. (15)–(20) [36,37].

The time (crank angle)-dependent expressions of in-cylinder pressure, burnt and unburnt gas temperatures, effective work, heat leak and heat loss are given respectively as:

$$\frac{C}{\omega}\left(\frac{V}{m}+\frac{\vartheta_1}{C_{p,b}T_b}\left(\left(x^2-x\right)(h_b-h_u)\right)\right)+\frac{h_{tr}}{\omega m}A_{\rm cyl}\begin{pmatrix}\sqrt{x}\frac{\vartheta_1}{C_{p,b}T_b}T_{\rm bw}+\\ (1-\sqrt{x})T_{uw}\frac{\vartheta_2}{C_{p,u}T_u}\end{pmatrix}$$
$$+\begin{pmatrix}\frac{\vartheta_1}{C_{p,b}T_b}(h_b-h_u)-\\ (\upsilon_b-\upsilon_u)\end{pmatrix}\frac{dx}{d\theta}+\frac{1}{m}\frac{dV}{d\theta}+\begin{pmatrix}\frac{\vartheta_1}{mC_{p,b}T_b}\begin{pmatrix}xh_b-\\ (1-x)h_u\end{pmatrix}-\\ H_u-\frac{V}{m^2}\end{pmatrix}\frac{dm_{\rm fi}}{d\theta}$$
$$+\frac{dp}{d\theta}=\frac{(1-x)\begin{pmatrix}\frac{\vartheta_1}{C_{p,b}T_b}\begin{pmatrix}-T_u\frac{ds_u}{d\theta}+\\ p\frac{d\upsilon_u}{d\theta}+\frac{du_u}{d\theta}\end{pmatrix}-\\ \frac{d\upsilon_u}{d\theta}+\frac{\vartheta_2}{C_{p,u}}\frac{ds_u}{d\theta}\end{pmatrix}\frac{d\phi}{d\theta}+x\begin{pmatrix}\frac{\vartheta_1}{C_{p,b}T_b}\left(p\frac{d\upsilon_b}{d\theta}+\frac{du_b}{d\theta}\right)-\\ \frac{d\upsilon_b}{d\theta}\end{pmatrix}\frac{d\phi}{d\theta}}{x\left(\frac{\vartheta_1}{C_{p,b}T_b}+\frac{\vartheta_3}{p}\right)+(1-x)\left(\frac{\vartheta_2^2}{C_{p,u}T_u}+\frac{\vartheta_4}{p}\right)} \tag{15}$$

where x, H_u, A_{cyl} are the burning fraction, lower heating value of fuel and heat transfer area of the cylinder. $C_{p,b}$, $C_{p,u}$; υ_b, υ_u; s_b, s_u; h_b, h_u are specific heat at constant pressure, specific volume, specific entropy and specific enthalpy for the burnt and unburnt zones respectively.

$$\vartheta_1=\frac{\partial\ \ln\ \upsilon_b}{\partial\ \ln\ T_b}\upsilon_b,\quad \vartheta_2=\frac{\partial\ \ln\ \upsilon_u}{\partial\ \ln\ T_u}\upsilon_u\quad \vartheta_3=\frac{\partial\ \ln\ \upsilon_b}{\partial\ \ln\ p}\upsilon_b,\quad \vartheta_4=\frac{\partial\ \ln\ \upsilon_u}{\partial\ \ln\ p}\upsilon_u$$

$$\frac{dT_b}{d\theta}=\frac{1}{C_{p,b}}\left(-\frac{h_{\rm tr}}{\omega m}A_{\rm cyl}\frac{1}{\sqrt{x}}T_{\rm bw}+\vartheta_1\frac{dp}{d\theta}\right) \tag{16}$$

$$\frac{dT_u}{d\theta}=-\frac{h_{\rm tr}}{\omega mC_{\rm pu}}A_{\rm cyl}\frac{T_{\rm uw}}{1+\sqrt{x}}+\frac{\vartheta_2}{C_{\rm pu}}\frac{dp}{d\theta}-\frac{\partial s_b}{\partial\phi}\frac{d\phi}{d\theta}\frac{1}{C_{\rm pb}} \tag{17}$$

$$\frac{dW_e}{d\theta}=-p\frac{dV}{d\theta} \tag{18}$$

$$\frac{dH_l}{d\theta}=\frac{Cm}{\omega}\left[(1-x^2)h_u+x^2h_b\right] \tag{19}$$

$$\frac{dQ_l}{d\theta}=\frac{h_{\rm tr}}{\omega}A_{\rm cyl}\left[\sqrt{x}T_{\rm bw}+(1-\sqrt{x})T_{\rm uw}\right] \tag{20}$$

The effective power and thermal efficiency are expressed as:

$$P_e=\frac{W_eN}{120} \tag{21}$$

$$\eta_e=\frac{P_e}{\dot{m}_fH_u} \tag{22}$$

where N is engine speed (revolution per minute).

Hohenberg [44] determines the coefficient of the heat transfer ($h_{\rm tr}$) as:

$$h_{\rm tr}=C_1V^{-0.06}p^{0.8}(xT_b+(1-x)T_u)^{-0.4}\left(\bar{S}_p+C_2\right)^{0.8} \tag{23}$$

where $\bar{S}_p$ is mean piston velocity in meters per second, C_1 = 130 and C_2 = 1.4 respectively. Sitkei [45] correlation is used to calculate ignition delay and it is written as following:

$$\tau_{\rm id}=0.5+0.1332p^{-0.7}e^{\frac{3.92782}{T}}+4.637p^{-1.8}e^{\frac{3.92782}{T}} \tag{24}$$

where p and T are average temperature and pressure of during the ignition delay. Dual Wiebe function states the burn fraction and x versus crank angle is used to express the heat release from the combustion and it is defined as [46]:

$$x=a_v\left[Q_{\rm pre}\left(1-e^{-a_v\left(\frac{\theta}{\theta_{\rm pre}}\right)^{(m_{\rm pre}+1)}}\right)+Q_{\rm dif}\left(1-e^{-a_v\left(\frac{\theta}{\theta_{\rm dif}}\right)^{(m_{\rm dif}+1)}}\right)\right] \tag{25}$$

where $Q_{\rm pre}$ and $Q_{\rm dif}$ are heat release rate of premixed and diffusive combustion. x is 0 at the beginning of the combustion and x becomes 1 at the end of the combustion. It can be rewritten by differentiating with respect to crank angle

$$\frac{dx}{d\theta}=a_v\left[\frac{Q_{pre}}{\theta_{\rm pre}}(m_{\rm pre}+1)\left(\frac{\theta}{\theta_{\rm pre}}\right)^{m_{\rm pre}}e^{-a_v\left(\frac{\theta}{\theta_{\rm pre}}\right)^{(m_{\rm pre}+1)}}+\frac{Q_{\rm dif}}{\theta_{\rm dif}}(m_{\rm dif}+1)\left(\frac{\theta}{\theta_{\rm dif}}\right)^{m_{\rm dif}}e^{-a_v\left(\frac{\theta}{\theta_{\rm dif}}\right)^{(m_{\rm dif}+1)}}\right] \tag{26}$$

$$\theta=\theta_r-\theta_b \tag{27}$$

where θ_r and θ_b are reference crank angle and start angle of combustion respectively, a_v is Wiebe energy converting factor, $m_{\rm pre}$; $m_{\rm dif}$ and $\theta_{\rm pre}$, $\theta_{\rm dif}$ are Wiebe form factors and combustion duration for the premixed and diffusive combustion conditions, respectively.

3. Theoretical model of steam injection and dual fuel mode

NO emissions are calculated by using extended Zeldovich mechanism taking into account 10 combustion products including (CO_2, H_2O, N_2, O_2, CO, H_2, H, O, OH, NO) [47,51]. The three reaction steps of NO formation is given in Table 1 and the rate constant is written as following:

$$k=A_AT^{B_A}e^{\frac{E_A}{T}} \tag{28}$$

The rate of NO formation (mol cm^{-3} s^{-1}) is given as:

$$\frac{d[\mathrm{NO}]}{dt}=\frac{2R_1(1-\alpha^2)}{1+\frac{\alpha R_1}{R_2+R_3}} \tag{29}$$

where $\alpha=\frac{[\mathrm{NO}]}{[\mathrm{NO}]_e}$ and $[\ ]_e$ stands for equilibrium concentration. The other constants written in Eq. (29) are expressed as following

$$R_1=k_{+1}[\mathrm{N_2}]_e[\mathrm{O_2}]_e=k_{-1}[\mathrm{NO}]_e[\mathrm{N}]_e \tag{30}$$

$$R_2=k_{+2}[\mathrm{O_2}]_e[\mathrm{N}]_e=k_{-2}[\mathrm{NO}]_e[\mathrm{O}]_e \tag{31}$$

$$R_3=k_{+3}[\mathrm{OH}]_e[\mathrm{N}]_e=k_{-3}[\mathrm{NO}]_e[\mathrm{H}]_e. \tag{32}$$

In this study the FARG and ECP codes which is developed by Olikara and Borman [48] is modified by adding steam injection and secondary fuel into the reactants by Gonca [37,42] for lean combustion circumstances. The combustion reaction used in the modified program is written as below:

For lower combustion condition;

$T\leqslant 1000$ K

$$\varepsilon\phi(\mathrm{C}_\alpha\mathrm{H}_\beta\mathrm{O}_\gamma\mathrm{N}_\delta+x\mathrm{C}_a\mathrm{H}_b\mathrm{O}_c\mathrm{N}_d)+[(0.21^*\mathrm{O_2})+(0.79^*\mathrm{N_2})]+y^*\mathrm{H_2O}\rightarrow\upsilon_1\mathrm{CO_2}+\upsilon_2\mathrm{H_2O}+\upsilon_3\mathrm{N_2}+\upsilon_4\mathrm{O_2}+\upsilon_5\mathrm{CO}+\upsilon_6\mathrm{H_2} \tag{33}$$

where, from chemical equation balancing for atoms:

$$\begin{aligned}&\upsilon_1=\varepsilon\phi(\alpha+x^*a)\\&\upsilon_2=\tfrac{\varepsilon\phi(\beta+x^*b)}{2}+y\\&\upsilon_3=\tfrac{\phi\varepsilon(\delta+x^*d)}{2}+0.79\\&\upsilon_4=\phi(-0.21)+0.21\\&\upsilon_5=0,\quad\upsilon_6=0.\end{aligned} \tag{34}$$

Table 1
Reactions of NO formation [51].

No.	Reaction	Forward/backward A_A (cm^3/mol s)	B_A	E_A (kcal/mol K)
1	N_2 + O ↔ NO + N	$7.6\times10^{13}/1.6\times10^{13}$	0/0	−38,000/0
2	O_2 + N ↔ NO + O	$6.4\times10^{09}/1.5\times10^{09}$	0/0	−3150/−19,500
3	OH + N ↔ NO + H	$4.1\times10^{13}/2\times10^{14}$	0/0	0/−23,650

$$x = \frac{M_d \rho_{sf} \upsilon_{\%sf}}{M_{sf} \rho_d \upsilon_{\%d}} \quad (35)$$

$$y = \frac{K_{\%} \varepsilon \phi (M_d + M_{sf})}{M_{ste}} \quad (36)$$

where m_d and m_{sf} are the masses of diesel and secondary fuel, M_{ste} M_d and M_{sf} are the total molecular weights of the steam, diesel and secondary fuel, ρ_d, ρ_{sf} and $\upsilon_{\%sf}$, $\upsilon_{\%d}$ are mass densities and percentage by volume of the diesel and secondary fuel, respectively. α, β, γ, δ ve a, b, c, d are atomic numbers of carbon, hydrogen, oxygen, nitrogen in diesel and secondary fuel, respectively. ε is molar fuel–air ratio and $K_{\%}$ is ratio of the steam mass to the total fuel mass

Where

$$\begin{aligned} M_d &= \alpha^* M_C + \beta^* M_H + \gamma^* M_O + \delta^* M_N; M_{sf} \\ &= a^* M_C + b^* M_H + c^* M_O + d^* M_N \end{aligned} \quad (37)$$

$$K_{\%} = \frac{m_{ste}}{m_d + m_{sf}} \quad (38)$$

$$\varepsilon = \frac{0.21}{\left((\alpha + x^* a) - \frac{(\gamma + x^* c)}{2} + \frac{(\beta + x^* b)}{4}\right)} \quad (39)$$

For higher combustion condition;

$T \geqslant 1000$ K

$$\begin{aligned} &\varepsilon\phi(C_\alpha H_\beta O_\gamma N_\delta + x C_a H_b O_c N_d) + [(0.21^* O_2) + (0.79^* N_2)] + y^* H_2O \\ &\rightarrow \upsilon_1 CO_2 + \upsilon_2 H_2O + \upsilon_3 N_2 + \upsilon_4 O_2 + \upsilon_5 CO + \upsilon_6 H_2 \\ &+ \upsilon_7 H + \upsilon_8 O + \upsilon_9 OH + \upsilon_{10} NO \end{aligned} \quad (40)$$

Where, from chemical equation balancing for atoms:

$$\begin{aligned} &\varepsilon\phi(\alpha + x^* a) = (y_1 + y_5)NY \\ &\varepsilon\phi(\beta + x^* b) + 2y = (2y_2 + 2y_6 + y_7 + y_9)NY \\ &\varepsilon\phi(\gamma + x^* c) + 2 \cdot 0.21 + y = (2y_1 + y_2 + 2y_4 + y_5 + y_8 \\ &+ y_9 + y_{10})NY \\ &\varepsilon\phi(\delta + x^* d) + 2 \cdot 0.79 = (2y_3 + y_{10})NY \end{aligned} \quad (41)$$

where NY is the total mole number and could be defined as follow:

$$NY = \sum_{i=1}^{10} \upsilon_i \text{ and } \sum_{i=1}^{10} y_i - 1 = 0 \quad (42)$$

$$2y_2 + 2y_6 + y_7 + y_9 - (y_1 + y_5) \cdot \frac{\varepsilon\phi(\beta + x^* b) + 2y}{\varepsilon\phi(\alpha + x^* a)} = 0$$

$$\begin{aligned} &2y_1 + y_2 + 2y_4 + y_5 + y_8 + y_9 + y_{10} \\ &- \frac{\varepsilon\phi(\gamma + x^* c) + 2 \cdot 0.21 + y}{\varepsilon\phi(\alpha + x^* a)} (y_1 + y_5) \\ &= 0 \end{aligned} \quad (43)$$

$$2y_3 + y_{10} - \frac{\varepsilon\phi(\delta + x^* d) + 2 \cdot 0.79}{\varepsilon\phi(\alpha + x^* a)} (y_1 + y_5) = 0$$

The mole fractions of the species are given with respect to y_3, y_4, y_5, y_6.

$$\begin{aligned} &y_7 = c_1 y_7^{1/2}, \quad y_8 = c_2 y_4^{1/2}, \quad y_9 = c_3 y_4^{1/2} y_6^{1/2}, \\ &y_{10} = c_4 y_4^{1/2} y_3^{1/2}, \quad y_2 = c_5 y_4^{1/2} y_6, \quad y_1 = c_6 y_4^{1/2} y_5 \\ &c_1 = \frac{K_1}{P^{1/2}}, \quad c_2 = \frac{K_2}{P^{1/2}}, \quad c_3 = K_3 \\ &c_4 = K_4, \quad c_5 = K_5 P^{1/2}, c_6 = K_6 P^{1/2} \end{aligned} \quad (44)$$

where K_i is the equilibrium constant and calculated by using Eq. (45).

$$\log K_i = A\, In\left(\frac{T}{1000}\right) + \left(\frac{B}{T}\right) + C + (DT) + (ET^2) \quad (45)$$

The A, B, C, D and E constants are taken from JANAF tables and these equations are solved with Newton–Raphson iteration method and results are obtained as in [43]. The charge pressure and temperature at the beginning of the compression stroke of the steam injected cycle are given as:

$$\begin{aligned} P_0 &= \frac{m_{ste} R_{ste} T_{ste} + m_a R_a T_a}{V} \\ T_0 &= \frac{m_a C_{\upsilon,a} T_a + m_{ste} C_{\upsilon,ste} T_{ste}}{m_a C_{\upsilon,a} + m_{ste} C_{\upsilon,ste}} \end{aligned} \quad (46)$$

where V is the cylinder volume which is changing with respect to crank angle, $C_{v,a}$ and $C_{v,ste}$ are the specific heat at constant volume, m_a and m_s are the masses, T_a and T_{ste} are the temperatures, R_a and R_{ste} are specific gas constants of the air and the steam in-cylinder at the beginning of the compression. R_a, R_{ste}, $C_{v,a}$ and $C_{v,s}$ are 0.287, 0.4615, 0.718 and 1.4108 kJ/kg K, respectively. Other values are changed with respect to cycle condition.

4. Results and discussion

A combustion simulation has been performed using two-zone combustion model so as to investigate the influences of steam injection on the performance and NO emissions of a diesel engine running with ethanol–diesel mixture. The combustion model have been performed and numerical results obtained from the results have been figured. The model has been conducted with the specifications of Superstar model engine given in Table 2, for the optimum steam rate (20% of injected fuel by mass) [33,38–40] with the 15% ethanol rate of injected fuel by volume and the results obtained from the model have been compared with each other. The results of two experimental and four theoretical engine modes have been investigated which are experimental (Exp-D) and model (Model-D) results of diesel engine, experimental (Exp-D + S20) and model (Model-D + S20) results of steam injected diesel engine, model results of diesel engine fueled ethanol–diesel mixture (Model-E15) and steam injected diesel engine fueled ethanol–diesel mixture (Model-E15 + S20). The properties of the blends are given in Table 3.

Fig. 1 shows the variation of in-cylinder pressures with crank angle for ethanol–diesel blend fuel and neat diesel fuel with and without steam injection. It is observed that ethanol–diesel blend causes to late ignition as ignition delay prolonged due to the lower cetane number compared to neat diesel. As seen in the figure, the

Table 2
Engine specifications.

Cylinder number	1
Injection	Direct injection
Stroke volume	0.92 dm^3
Engine speed range	1200–2400 rpm
B	10.8 cm
S	10 cm
R	17
Power	13 kW
RGF	0.01
θ_{di}	8 CA
θ_{si}	35 BTDC
n	2
The range for ϕ	0.8–0.85
p_1	1 bar
T_1	300 K
T_{ste}	406 K
T_w	450 K

Table 3
The properties of the mixtures.

	Diesel [40]	Ethanol [23]	15% eth.–dies.	D	D + S20	E15	E15 + S20
Diesel (%vol)	–	–	–	100	100	85	85
Ethanol (%vol)	–	–	–	0	0	15	15
Steam (%mass)	–	–	–	0	20	0	20
Chemical formula	$C_{14.4}H_{24.9}$	C_2H_5OH	–	–	–	–	–
Mole weight (g)	198	46	–	–	–	–	–
Density (g/cm^3 at 20 °C)	0.84	0.789	0.828	–	–	–	–
Lower heat value (MJ/kg)	42.5	27	39.5	–	–	–	–
Cetane number	45	8	–	–	–	–	–

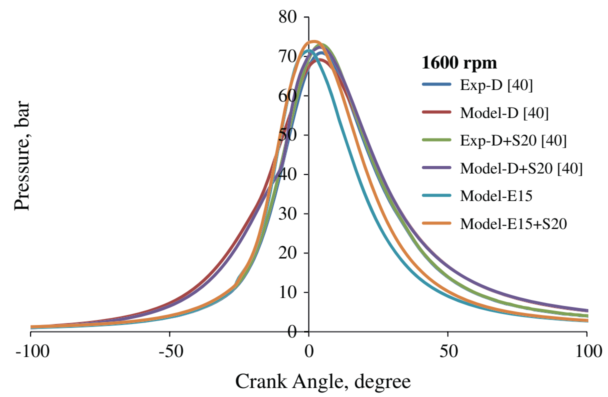

Fig. 1. Variation of in-cylinder pressure with crank angle for different engine modes at 1600 rpm.

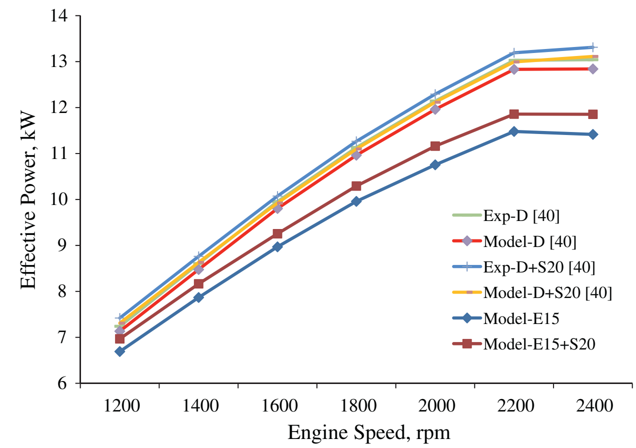

Fig. 3. Variation of effective power with engine speed for different engine modes.

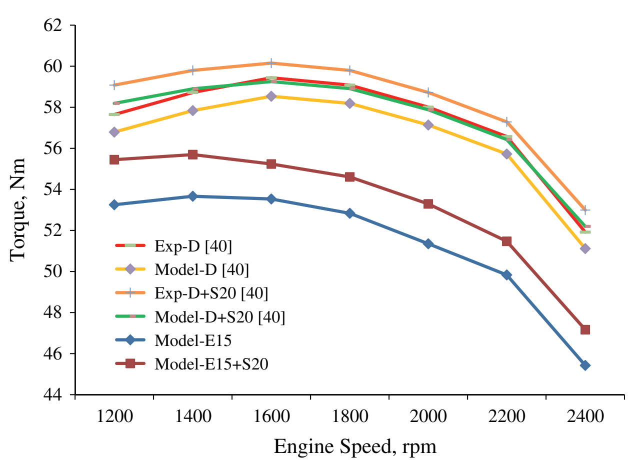

Fig. 2. Variation of torque with engine speed for different engine modes.

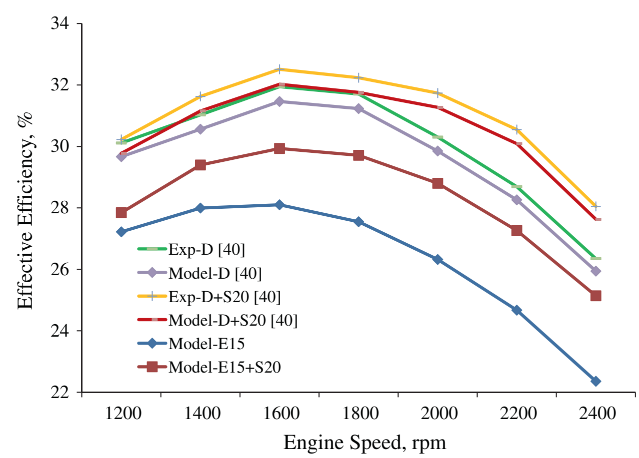

Fig. 4. Variation of effective efficiency with engine speed for different engine modes.

maximum combustion pressure is increased with steam injection and the diesel engine running with ethanol–diesel blend. However, when the steam injection applied to the engine the maximum in-cylinder pressure more increased owing to better atomization and shorter ignition delay and a considerable increase is seen in the engine performance.

The engine torques and effective powers of different engine modes with respect to engine speed are depicted in Figs. 2 and 3 respectively. As could be seen from the figures, there are reductions in the torque and effective power of the diesel engine running with ethanol–diesel blend. The heating value of fuels burned in the engines influences the torque and power of an engine. Hence, without any modifications, ethanol–diesel fuel blends leads to reductions in those due to their weak energy contents compared to neat diesel fuel [1–12]. This is one of the reasons of the reductions in the torque and power. Other reducing factors are lower density and cetane number of blends. The prolonged ignition delay causes to a deterioration in the engine performance [1,49]. In the ethanol–diesel blend operating mode, the maximum reductions in the torque and effective power are 11% at 2400 rpm in the comparison between the model results of neat diesel and ethanol–diesel blend modes. However, when steam injection applied to the engine the maximum reductions in the torque and effective power could be decreased to 7.6%.

Fig. 4 illustrates the variation of effective efficiency with respect to engine speed for different engine modes. The effective efficiency stands for how efficiently the fuel energy is transformed to mechanical output. As can be seen in the figure, the effective efficiency of the engine fueled with blend is less than diesel alone. The

reasons of this reduction are ethanol–diesel blend has much lower heating value than pure diesel and lower useful energy conversion rate [12,23,24]. The maximum reduction in the effective efficiency is 13.2% at 2400 rpm in the comparison between the model results of neat diesel mode and mixture mode. On the other hand, when steam injection implemented to the engine the maximum reduction in the effective efficiency could be reduced to 3.1%.

NOx formation is strongly dependent on peak combustion temperatures, oxygen concentrations and residence time [23,24,30]. There are 34% oxygen in ethanol and its cetane number is very low in comparison with diesel fuel. Premixed combustion is prolonged due to lower cetane number and so the combustion temperature is increased. However, the lower heating value of ethanol is less than diesel fuel and latent heat of vaporization of ethanol is higher than diesel fuel, these properties of ethanol decreases peak temperature in the cylinder [23,24,26]. It is seen from Fig. 5 that cetane number and oxygen concentration are more dominant than lower heating value and latent heat of vaporization on account of raising peak combustion temperature in the engine cylinder. Thus, as can be seen in Fig. 6, NO emissions enhanced with the ethanol–diesel blend as compared to neat diesel because of lower cetane number, higher oxygen concentration and ignition delay. Application of steam injection into the diesel engine positively affects the combustion efficiency and considerably decreases NOx formation. When steam is injected into combustion chamber, water droplets have good atomization and vaporize rapidly absorbing the heat of the cylinder charge owing to its high heat capacity and partial pressure of oxygen increases. In conclusion, peak combustion temperature and thus, NOx emissions decreases [50]. As could be observed from Figs. 5 and 6, steam injection leads to the reduction in NO emissions compared to modes without steam injection at all engine speeds, because the peak combustion temperatures decrease. The maximum NO is 846 ppm in the engine with running ethanol–diesel blend at 1400 rpm. The minimum NO is 498 ppm at 2400 rpm with steam injection.

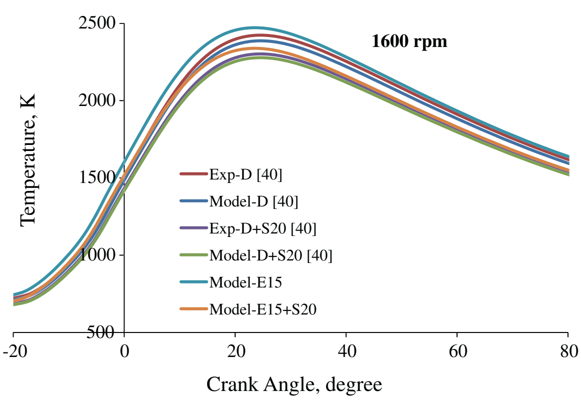

Fig. 5. Variation of in-cylinder temperature with crank angle for different engine modes at 1600 rpm.

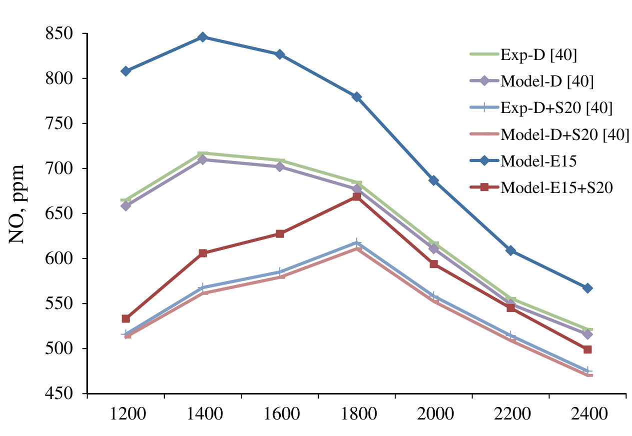

Fig. 6. Variation of NO emissions with engine speed for different engine modes.

5. Conclusion

In this study, the effects of steam injection on the performance and NO emissions of a diesel engine fueled with ethanol–diesel fuel have been simulated by two-zone combustion model. When the modes of diesel engine fueled with ethanol–diesel fuel compared with those of pure diesel fuel, a considerable increment in NO emissions and a reduction in the torque, effective power and effective efficiency have been observed. However, NO emission is decreased and performance is improved when steam injection is applied into the diesel engine. The effective efficiency and effective power of the engine fueled mixture is increased up to 12.5% and 4.1%, respectively, NO emissions reduced up to 34% with steam injection. In conclusion, application of the steam injection into the diesel engine gives the best results in terms of NO emissions and performance. Thus, this technique could be implemented to the diesel engines in order to increase the usability of ethanol as a diesel engine fuel.

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